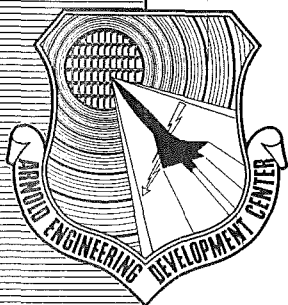AEDC-TDR-62-168

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FATIGUE TESTS AND CRACK DETECTION ON SOME TYPES OF AXIAL-FLOW COMPRESSOR ROTOR BLADES

By

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ARO, Inc.

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NOTICES

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a subsidiary of Sverdrup and Parcel, Inc.

ABSTRACT

Fatigue data were obtained from several types of compressor rotor blades that are in use in axial-flow compressors at the von Kármán Gas Dynamics Facility. The data provided a general comparison of "as manufactured" endurance limits and were used as a verification control in evaluating a supplementary method of detecting failed blades during systematic maintenance inspections.

By the presentation of stress versus cycles to failure curves, a comparison is made between AISI number-8630 cast steel blades as originally furnished with the compressors, AISI number-8630 cast steel shot-peened blades, and AISI number-403 forged, heat-treated and machined blades.

A method is presented which may be used to supplement the magnetic particle method for detecting cracked rotor blades.

PUBLICATION REVIEW

This report has been reviewed and publication is approved.

Darreld K. Calkins
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Colonel, USAF
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1.0 INTRODUCTION

Semiannual maintenance inspections of the axial-flow compressors in the von Kármán Gas Dynamics Facility (VKF), Arnold Engineering Development Center (AEDC), Air Force Systems Command (AFSC), USAF, revealed that the rotor blades were developing random cracks near the fillet radius. The cracks were determined to be fatigue cracks resulting from vibratory stresses imposed during compressor operation.

The designer and manufacturer of the compressors proposed to furnish at an increased cost a redesigned rotor blade of heavier construction made of AISI type-403 forged, heat-treated, and machined steel. The redesigned blade would have lower operating stresses and higher fundamental frequencies of vibration than the original type-8630 cast steel blades.

Although the relative order of endurance strengths of the various materials could be predicted, it was of interest to determine the "as manufactured" endurance limits of the blades to assure that other factors had not affected the anticipated increase in life. Also, the fatigue data would provide a control basis for evaluating the effectiveness of a method of detecting failed blades. Consequently an investigation was made to determine:

1. natural frequencies and endurance life of the type-8630 cast blades and the type-403 forged, machined and heat-treated blades

2. the effect of shot-peening on the endurance life of either type blade and

3. a method of detecting failed blades to supplement the magnetic particle test.

2.0 APPARATUS

2.1 TEST BLADES

The blades tested were of the following type rotor blades taken at random from existing stock:

1. AISI type-8630 cast steel blades as originally furnished with the compressors

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2. AISI type-403 steel forged, heat-treated, and machined as received from the manufacturer and

3. types-8630 and 403 blades shot-peened to a height of 2-1/2 in. from the root on both sides, P-23 shot, arc height 0.010 to 0.020 (Alman "A" strip and number 2 gage).

The physical characteristics of the type-8630 and type-403 blades are given in the appendix.

2.2 TEST STAND

A stand to support the blade during fatigue testing was fabricated and bored to duplicate the mounting of a blade in its rotor (Fig. 1). The weight of the stand was approximately 125 lb, which was sufficient to provide a stable base when the blade was vibrated.

The coil used to transmit a magnetic force to the blade to excite vibrations was mounted to provide vertical and horizontal positioning relative to the blade. A probe was mounted on the opposite side of the blade to determine the tip amplitude of the blade. The test stand was placed on a one-inch-thick section of acoustical tile to reduce the amount of vibration transmitted to the test bench. The entire stand was placed inside a horizontally positioned telephone booth to reduce the noise level in the working area.

The coil core was constructed of type-8630 steel laminations, 0.031-in.-thick, forming an "E" shaped core with base dimensions of 2-3/4 in. by 1 in.; each leg length was 3-1/2 in. long and 3/8 in. wide. Two coils consisting of 1000 turns of gage number 22 varnished copper wire were wound on 1/8-in.-thick micarta forms. The coils were placed on the two outer legs of the core and fixed with Armstrong Al cement. The two coils were connected electrically parallel to the frequency generator. Each coil when completed had a d-c resistance of eight ohms.

A depth micrometer was modified and mounted on the test stand on the side of the test blade opposite the vibrating coil. The probe end of the micrometer assembly was made of tungsten to better withstand the destructive effect of an electric arc between the probe and the blade.

Components and connections for the fatigue tests are shown in Fig. 2. The apparatus and connections used to obtain natural frequency data of installed rotor blades are shown in Fig. 3.

3.0 PROCEDURE

3.1 STRESS VERSUS CYCLES TO FAILURE TESTS

To vibrate the blade, the coil was positioned as close to the blade as possible, yet far enough away to provide a clearance between the blade and coil when the blade was vibrating. Approximately two microfarads were connected in series with the coil to approximate series resonance. The test components and connections are given in Figs. 2 and 3.

Approximately one-half of the fundamental frequency of the blade was set as the output of the frequency generator, and the power level of the generator was set to produce a low amplitude of vibration. Alternate adjustment of generator frequency and circuit capacitance was made until maximum blade amplitude was obtained for the fixed power setting of the generator. Under these conditions the output frequency of the generator was one-half of the fundamental frequency of the blade, and the coil circuit was tuned to series resonance, producing maximum power transfer to the coil. The power output of the generator was then increased until the desired blade tip amplitude was obtained. Under these conditions the blade was vibrated until completion of the test.

At the start of these tests the stress amplitude was monitored with strain gages and a recorder. The strain gages were cemented to both sides of the blade and located arbitrarily approximately 3/4 in. from the root. The strain-gage apparatus was calibrated by hanging weights from the tip of a horizontally mounted blade and correlating weight, tip deflection, location of strain gage, and recorder trace. Vibrating stress levels were established according to recorder indications, and the cycles to failure or cycles to completion of the test were determined by the time lapse of the recorder chart.

The strain-gage method was discontinued because of strain-gage failures, magnetic field interference, and time consumed in calibrations. Strain-gage failures occurred as breakage of the wires within the gage, presumably as a result of fatigue. When failure of the gage occurred, the test was continued by monitoring the remaining gage; however, frequently both gages would fail at approximately the same time, and installation of new gages and recalibration was necessary. The strain gages were sensitive to the fluctuating magnetic field of the vibrating coil which introduced errors to the indicated stress levels. Also, the calibration of the apparatus was affected by the temperature of the blade, which increased during the first hour of operation.

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To avoid the difficulties associated with strain gages, the spark gap apparatus, shown by Figs. 1 and 2, was developed from ideas presented in Ref. 1. Blade tip amplitude was established by setting the output of the power supply to 500 volts, advancing the probe to the vicinity of the blade until an electric arc occurred, and then setting a value on the micrometer corresponding to the tip deflection desired. The blade was then excited by raising the output power of the frequency generator until an arc was produced. It is reported in Ref. 1 that the arc so produced is approximately 1/100 millimeter. The method used to compute the tip deflection for a given reference stress is given in the appendix.

After startup, the apparatus was monitored constantly for approximately two hours to make corrections in generator frequency and power to compensate for blade warmup and equipment drift resulting from warmup. After equipment stabilization, the tip amplitude probe voltage was removed to prevent excessive erosion of the blade at the point of arcing. Probe voltage was then switched on at 15 to 30-min intervals to check that the amplitude was constant. The blade was excited until vibration at the set amplitude and frequency could not be sustained even with an increase in generator power output. A decrease in amplitude accompanied the occurrence of a crack because as the crack developed, the natural frequency of the blade decreased, the coil circuit became detuned to the new frequency, and a loss of transferred power to the coil resulted.

When a drop in frequency of approximately 2 cps was detected, the blade was subjected to the magnetic particle test (MPT, Ref. 2) to verify the crack. If the defect could not be found with the MPT at this time, the blade was further vibrated at its new frequencies to aggravate the crack until it was detected by the MPT. The time when the initial frequency drop was detected was taken as a point for the S-N curve. Total cycles to failure of the blade was taken as twice the counter reading and was also computed from the time lapse of the test. From the data thus obtained, stress versus cycles to failure curves were prepared for type-8630 (Fig. 4), type-8630 shot-peened (Fig. 5), and type-403 (Fig. 6) rotor blades.

3.2 DETECTION BY FREQUENCY CHANGE

Figure 3 indicates schematically the method used to determine the fundamental frequency of blades while they were installed in the compressor rotor. The same method was employed in the laboratory.

The blade was struck with a plastic hammer, and the vibrations produced were detected by the transducer, amplified, and applied to the y-axis input of the oscillograph. While the blade was vibrating,

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the output frequency of the function generator, which was applied to the x-axis input of the oscillograph, was adjusted to obtain a Lissajous circle on the oscillograph. A Lissajous circle was produced as a result of blade frequency being equal to the function generator frequency. The function generator frequency was displayed by the counter, and this value was recorded as the fundamental frequency of the blade.

Fundamental frequency data were taken from the axial rotor blades during five semiannual plant maintenance periods. Frequency readings of each blade were compared to previous readings for the same blade from the second maintenance period on. If a blade showed a decrease in frequency more than the trend of the majority of other blades in the same row, the blade was removed and subjected to the magnetic particle test (MPT). If the MPT revealed a crack, the blade was discarded as being defective. If a crack was not revealed by the MPT, it was returned to the laboratory for fatigue testing.

When the blades were returned to the laboratory, they were vibrated at a stress level well above the endurance limit shown by the S-N curves. A record was kept of variance of fundamental frequency with the total cycles of vibration. The data were plotted as frequency versus cycles for each blade so tested.

When the fundamental frequency of the blade under test decreased by approximately two cycles per second, the blade was checked by the MPT. If the MPT failed to detect the crack, the fatigue test was resumed. If the MPT detected the crack, it was considered that the crack had formed as the frequency was decreasing. When the crack was so verified by the MPT, a comparison was made between the total cycles to failure of the blade and the appropriate stress versus cycles curve of Figs. 4, 5, and 6. If the cycles to failure value compared favorably with the curve, it was considered that the blade had been sound when removed from the compressor. Otherwise, it was considered that the frequency drop shown by the field data represented a crack.

4.0 RESULTS AND DISCUSSION

4.1 STRESS VERSUS CYCLES TO FAILURE DATA

Stress versus cycles to failure data are presented for each type of blade that has been used in VKF (Figs. 4, 5, and 6). The three curves were obtained to provide a comparison of the operating endurance limits of the blades and to form a basis for verification of failures when used blades were fatigue tested.

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In addition to and including the reasons for the usual scatter associated with fatigue of full-scale factory-run items, the scatter of the data may be attributed to the manner in which the reference stress was taken. The reference stress, as described and calculated in the appendix, was taken at the same place on all blades and is not necessarily the stress at the point of failure. The point of failure would be determined by surface preparation, nicks, scratches, sharp edges of notches, blade geometry, voids, etc., that cause stress risers which may not be at the location of the reference plane where the stress was calculated. In fatigue tests, the scatter is reduced proportionally as the test specimens are more identically prepared. Since the blade specimens used here were not prepared to make them uniform and were tested as they were manufactured, scatter becomes significant. Such is desirable, because results are indicative of what may be expected when the blades are in practical use, and a better indication of the fatigue life of a compressor blade is provided than when the specimens are perfectly prepared. The same factors that caused the start of fatigue cracks during the tests will also be predominant in causing fatigue cracks while in compressor use.

The conditions were taken as they were so the data would more nearly reflect the conditions of blades in use.

4.2 COMPARISON OF FATIGUE DATA

The data obtained indicates that the type-403 and type-8630 shot-peened blades have higher fatigue strengths than the original type-8630 blades. The most significant improvement is shown to be in the type-403 blade which has an indicated endurance limit of 44,000 psi as compared to an indicated endurance limit of 27,000 psi for the type-8630 blade.

An increase of approximately 4000 psi in endurance limit is reflected as a result of shot-peening the type-8630 blades. A comparison of the curves shows that shot-peening improved the fatigue strength during the high stress part of the curve but that much of the beneficial effects were lost after prolonged cyclic stress.

The great amount of scatter of the data for shot-peened type-403 blades indicates that some of the type-403 blades were improved by shot-peening, whereas others were unaffected. Where an improvement of fatigue life in a given blade is indicated, it may be attributed to the results of shot-peening having removed some sharp edges and other stress raisers and may have smoothed out some roughness of machining. Shot-peening appears to offer unpredictable results when applied to a type-403 blade.

Of the types of blades tested, the type-403 blade offers the longest operating life because of its greater fatigue strength.

4.3 DETECTION OF CRACKED BLADES

Observations during fatigue tests confirmed that a decrease in blade frequency accompanied the development of a crack. On this basis, attempts were made to detect failed or failing blades by noting the difference between successive frequency determinations during five semiannual plant inspection periods.

The success of such detection depended on controlling or accounting for all factors which would naturally change the frequency of the blades. Practically, such factors as ambient temperature could not be controlled. Therefore, the frequency differences were considered statistically so a normal variation of differences could account for the uncontrolled effects of temperature, accuracy, etc. With a normal grouping of frequency differences, a blade which was in the process of failing or undergoing an undesirable change, such as loosening of the fastening nut, the frequency decrease between successive investigations would be significantly outside the grouping of the distribution. Blades removed during the March 1961 maintenance period exhibited frequency decreases greater than the negative values of Fig. 8.

To illustrate the procedure followed to determine that a particular blade should be investigated for cracks, the following actual case for compressor No. 14, row 1, type 8630 blades is given.

The frequency of each blade in row 1 was taken and recorded during September 1960, and the frequency of each blade in row 1 was taken and recorded again during March 1961. The two frequencies of each blade were compared to obtain the difference. A positive sign was assigned to the remainder if the blade frequency had increased, and a negative sign was assigned to the remainder if the frequency of the blade had decreased. The following table of five blades will serve to show how this was accomplished for all blades in row 1.

| | Frequency, cps | | Frequency |
| | 9-60 | 3-61 | Difference, |
Blade No.	Inspection	Inspection	cps
1	236.3	236.8	+ 0.5
2	234.2	234.3	+ 0.1
3	242.1	242.2	+ 0.1
4	240.8	240.8	0.0
5	234.4	234.3	- 0.1

When all 50 blades in row 1 were so tabulated, the differences were grouped numerically as in the following example:

Number of Tabulated Differences
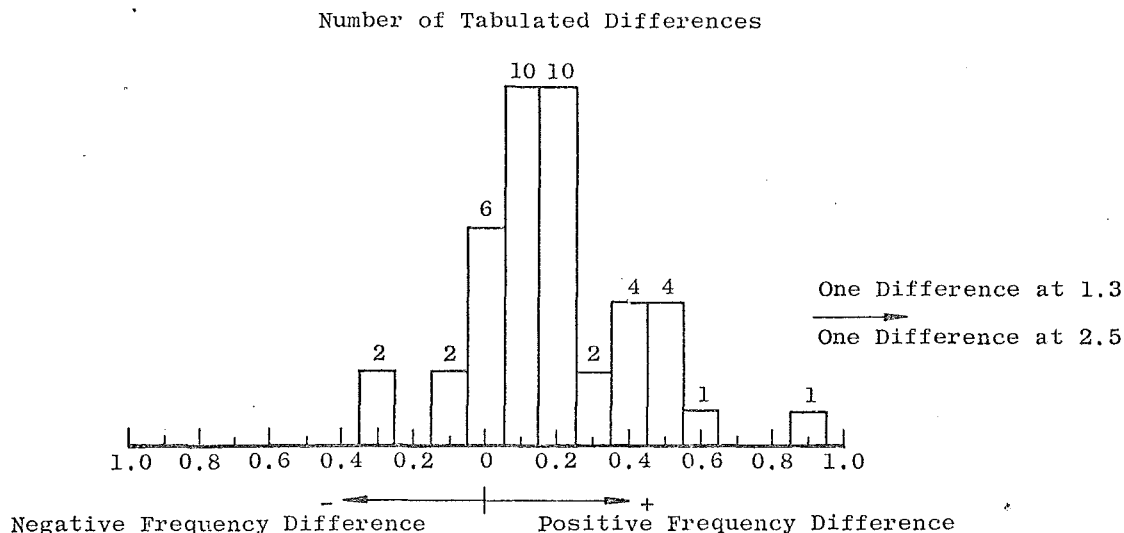
This graph represents only 44 blades because six of the blades had recently been changed to type-403 steel for reasons not connected to this illustration. The grouping of this range of differences is between -0.1 cps and +0.6 cps, and since the grouping is not concentrated about 0 cps, we assume that some or all reasons that may cause the natural frequency of a blade to change have influenced a general increase in frequency. This assumption includes a shift of instrumentation zero as well as natural changes to the blade. The three frequencies to the right of the grouping probably can be explained as an error in determining or recording the frequencies; it is discussed under section 4.5 of this report how these errors are eliminated by successive frequency determinations. In any event we have no basis for suspecting a defective blade by the indication of an increasing (positive) frequency change, and the 1.3-cps and 2.5-cps differences were ignored until future determinations could locate the error.

The two blade frequencies shown on the graph at -0.3 cps apparently exhibit decreasing frequencies outside the normal grouping, so we assume that the decreasing frequency change greater than all others in the row being considered may represent a cracked blade. Or in fact, these two blades should be inspected with all means available such as the MPT to determine if crack indications or other detrimental factors are present. In this actual case the two blades were pulled and subjected to MPT, X-ray inspection, and fatigue test in an attempt to locate cracks. One of the two blades was determined to be defective by fatigue tests, the

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history of which is given as blade no. 276 in Fig. 7. The other blade exhibiting a 0.3-cps decrease was not proved to be defective.

This example is given for row 1 of compressor no. 14 and is further represented as the first bar of the graph of Fig. 8. Figure 8 also shows a bar for the range of frequency differences of every row of each axial compressor in VKF. The graph of Fig. 8 shows such random behavior of frequency differences between row numbers that it appears impossible to fix any universal frequency change as a criteria for suspecting a defective blade.

It was considered that the S-N curves of Figs. 4, 5, and 6 represent the fatigue life of a normal compressor blade as used in VKF. If a suspected blade failed to exhibit a fatigue life comparable to the curve, it was considered to have been in the process of failing when removed from the compressor. Conversely, when the fatigue life of a suspected blade compared favorably with the curve, it was considered to be a sound blade if other methods could detect no failure.

The degree of agreement between the fatigue life of a suspected blade and the controlling S-N curve may be seen by comparing Fig. 7 with the proper S-N curve. Figure 7 shows the fatigue test history of some representative type-8630 blades, all of which cracked during the test (with the exception of blade no. 288), and the crack was positively detected at the terminus of the frequency versus cycles to failure curve. When a blade such as no. 263 was compared to the S-N curve of Fig. 4, it was expected from the S-N curve that a failure should occur at better than 800,000 cycles at the stress level of 32,500 psi, whereas the history plot of Fig. 7 shows that the frequency of the blade decreased 5 cps during the first 300,000 cycles of testing. Since the S-N curve of Fig. 4 was compiled by plotting the point at which the blade made a sharp decrease in frequency rather than when the crack was discovered by conventional means, the comparison shows that the test blade was failing as soon as the fatigue test started. The failure point of all the blades of Fig. 7 (except 288) was considered to be at the start of the test. This is verified by the rates of frequency decrease at the beginning of the test which are greater than expected as a result of temperature increases.

Blade no. 286 of Fig. 7 does not show a rapidly decreasing frequency initially, but the rate of frequency change became larger as the test proceeded and did not become stable in frequency after warm-up as with sound blades. This blade gives the impression that failures were occurring slowly at the start of the test and most certainly between 600,000 and 800,000 cycles.

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The other blades shown in Fig. 7 follow the characteristics of blade no. 263 initially, which is an indication of when the failures started; however, the manner in which the frequency decreased, and during certain periods did not decrease, may be an indication of the rate of growth of fatigue cracks. For example, blade no. 263 exhibited a stable frequency between 300,000 and 500,000 cycles and there after decreased until a large surface crack was detected. As a comparison, blade no. 276 never exhibited a stable frequency while being tested. The stable frequency periods of the plots may be periods when cracks are not growing, and the remaining section of the blade is being fatigued as if it were a blade of reduced cross section.

Included in Fig. 7 is the history of blade no. 288 which has a compressor history of a decreasing frequency and fatigue test of a decreasing frequency until at two million cycles the frequency became fixed and did not change even to eight million cycles. A crack was not found on this blade by the MPT. X-ray inspection revealed inclusions near the tip of the blade, which is not uncommon for the type-8630 blades and is not necessarily a proof of internal cracking. Although blade no. 288 is the only case of its kind that was experienced during this test, its history is included to show what may be one limitation to the frequency method. This blade has shown a severe decrease from the initial frequency, yet its frequency became stabilized and the expected life was not decreased.

Another limitation of the frequency method is shown by the shape of curves 263 and 285 of Fig. 7. Here the frequency starts decreasing, becomes constant for a period of time, and then continues to decrease. If two comparative frequency data were taken during the time the frequency was constant, no differences would be noted, and yet the blade would be cracked. Under these circumstances, the blade would not be detected by the frequency method. However, this limitation would not apply where periodic frequency readings are started with installation of the blade.

The effect of temperature was necessarily considered when interpreting the curves of Fig. 7. For rows 1 through 6 blades, an approximate 1-cps decrease in frequency during the first million cycles was attributed directly to the increasing temperature of the vibrating blade. For rows 7 through 9 blades, the temperature of the blade increased more during the first million cycles, and according to observation, the frequency decrease was approximately 2 cps. After the first million cycles, the blade temperature would stabilize, as would the frequency. A crack was detected by noting a frequency decrease rate greater than could be attributed to the temperature effect.

Defective blading, other than blades with surface fatigue cracks, was discovered by the frequency method during this investigation. The following conditions would otherwise have been discovered only accidentally with existing methods.

1. Blades that had not been fastened securely during installation exhibited lower frequencies and faster vibratory damping when checked.

2. Significant deviations in blade dimensions were found. A shorter, thicker, or narrower blade would have a frequency higher than normal, and a longer, thinner, or wider blade would have a frequency lower than normal. Mass distribution affecting blade geometry influences the frequency of the blade.

3. One type-403 blade was found on the second frequency check to have a large frequency decrease. X-ray examination revealed that the blade had internal voids which were being connected by cracks during operation.

4. A rotor which had been re-bladed recently with type-403 blades was frequency checked and was found to contain six type 8630 blades. This was detected by the large difference in frequency of the two types of blades.

4.4 RESULTS OF DETECTION METHODS

The following tabulation lists all blades that were removed from the six VKF compressors during three maintenance periods because of suspected cracks. The number of blades inspected at each period is different because during the March 1961 period one compressor rotor was changed from no. 8630 to no. 403 blades. An inspection of both the old and new rotor blades was required. During the October 1961 period two rotors were re-bladed. This left all six compressors complete with type-403 blades, so the April 1962 inspection was of type 403-blades entirely. At each inspection, all blades were inspected by both the MPT and frequency method; the number of blades removed by each method and the number of these that were verified to be failed are tabulated in the table on the following page.

Date of Inspection	March 1961				October 1961				April 1962			
Type of Blade	No. 8630		No. 403		No. 8630		No. 403		No. 8630		No. 403	
Total Blades Inspected	1800		1350		1250		2250		0		2700	
Method of Detection	Freq.	MPT	Freq.	MPT	Freq.	MPT	Freq.	MPT	Freq.	MPT	Freq.	MPT
Total Blades Removed	34	22	4[a]	1	0	0	0	0	0	0	2[b]	0
Total Verified	10	9	1[a]	0	–	–	–	–	–	–	1[b]	–

(a) Two of these four blades are 9th row blades and are too stiff to be fatigue tested with our existing apparatus. Other tests for cracks were negative. X-ray inspection revealed that the one verified blade had inclusions and cracks.

(b) The blade listed as verified was detected at the first frequency check of a type-403 bladed rotor. It was suspected because of an indicated frequency which was approximately 100 cps lower than the average of all other blades in the same row. Measurements revealed the blade to have dimensions significantly different from all other like blades. The dimension difference was the cause of the lower frequency. The other blade was not verified.

All of those blades listed in the tabulation as removed by the MPT during the March 1961 inspection had original MPT indications in or near the notch and fillet area. Those listed as verified were done so by either laboratory fatigue tests as described for the frequency method verifications or by laboratory MPT.

As indicated, both the MPT and frequency method caused removal of sound blades. Computed on the basis of number of blades fatigue tested or otherwise proved defective, the frequency method ratio of verified blades to blades removed is 12/38 = 0.315, whereas the MPT ratio is 9/23 = 0.391. These figures ignore two blades removed for frequency decreases which were not fatigue tested. It is expected that the ratio for frequency check removals will be higher as experience dictates to the interpreter that a wider range of frequency differences may be tolerated. Also, the more history available on a blade, the more intelligently a decision to remove can be made. An indication of these factors is shown by the low number of blades removed during the April 1962 inspection.

4.5 ACCURACY

Inaccuracies in the method of detecting blade frequencies occur predominantly from the operator in determining the quality of the Lissajous circle. Obtaining a stable circle is an art requiring careful judgment of the operator, and any error in determining a stable circle is reflected in the final frequency determination. Inherent inaccuracies of the counter and function generator are considered minor relative to operator adjustments. Inasmuch as the decision to remove a suspected blade was made on the basis of a frequency difference which was outside the range of all other differences in the row being checked, equipment inaccuracies were minimized.

The range of frequency differences between the November 1960 and March 1961 maintenance periods are presented in Fig. 8 and the following tabulation:

Row	Compressor	Maximum Difference, cps	Maximum Difference as Percent of Average Frequency
1	11	2.1	0.9
2	21	2.4	0.7
3	13	1.1	0.3
4	21 & 22	1.1	0.2
5	21	1.7	0.3
6	21	1.8	0.2
7	22	1.6	0.2
8	13	4.3	0.4
9	13	4.7	0.4

These differences are smaller than the differences between the first two maintenance periods because of errors introduced during the first period. Considerable error was introduced during the first data run because of a defective counter which was influenced by operation of nearby electrical machinery.

The data show that readings were repeated to within one percent of previous readings. This value includes the effect of natural changes which may be inherent in the blade and its method of support, such as an increase in fundamental frequency after installation. The data does reflect a trend toward a positive frequency difference between successive readings immediately following initial installation especially.

Errors in first frequency determinations are most difficult to detect because only one determination makes correlation impossible. The

second determination may be compared to the first, and thus the blade is shown to be a failure, or a determination is shown to be in error. The third determination should establish a trend of the blade frequency or produce reasonable agreement between two of the determinations whereby the erroneous determination can be detected. Should the third determination be lower than the others, the blade may be a failure or an error may have been made in the third determination, but in such an event, the third determination can be repeated to bring it in agreement to assure that an erroneous frequency has not been recorded. As more frequency determinations are made, less chance will exist that an erroneous determination will effect the removal of a blade.

4.6 TORQUE OF FASTENING NUT

It was found that a change in torque applied to the fastening nut of a blade caused a corresponding change in fundamental frequency. For example, with one blade a change in torque of from 1200 in.-lb to 1800 in.-lb changed the natural frequency from 342.8 cps to 343.6 cps.

Unpublished work by the Rocket Test Facility's Plant Operations Branch, ARO, Inc., reveals that the stress in the fillet area of a certain blade increased proportionally with torque from zero torque to 1-1/2 times design torque. Therefore, the increased stress resulting from increased torque apparently produces an increased fundamental frequency.

The curves of Fig. 7 reflect this effect where there is a difference between the compressor-installed frequency and the test bench frequency.

4.7 RANGE OF BLADE FREQUENCIES

Considerable fundamental frequency variations were found to exist between blades of the same row number.

Ideally, operating conditions which would cause excitation of blades at their fundamental or certain harmonic frequencies are avoided. To show the range of frequencies that would require consideration here, the following is presented:

Blade Static Fundamental Frequency, cps					
Row Number	Computed Frequency*		Average of 50 Blades		Range of Averaged Frequencies	
	Type 8630	Type 403	Type 8630	Type 403	Type 8630	Type 403
1	233	299	236.9	302.7	228.4-251.4	297.7-308.9
2	277	351	279.0	363.1	266.9-288.4	356.4-372.4
3	334	423	333.5	435.0	323.6-342.0	424.5-442.6
4	402	516	401.2	533.9	391.4-414.2	524.3-540.2
5	480	624	497.0	661.4	462.6-521.1	652.5-671.4
6	563	710	566.0	748.2	534.7-584.5	733.6-758.3
7	651	815	663.3	855.7	631.1-683.5	832.9-871.5
8	755	927	773.9	999.5	747.5-802.7	975.8-1014.3
9	912	1124	910.2	1179.2	851.1-932.6	1158.2-1198.8

*The computed frequencies were supplied by the Engineering Analysis Section, VKF.

The tabulation shows that there is a considerable variation in fundamental frequency between these blades which are otherwise identical within manufacturing and installation tolerances.

5.0 CONCLUDING REMARKS

Fatigue tests were conducted to determine the relative endurance life of three types of complete rotor blades, and the S-N curves obtained were used as a control in determining the effectiveness of a method to detect cracked or defective rotor blades. The endurance limits were determined to be equal to or better than the following values:

Blade Types	Endurance Limit, psi
AISI 8630 cast steel	27,000
AISI 8630 shot-peened	31,000
AISI 403 forged steel	44,000
AISI 403 shot-peened	Inconclusive

The method of detecting cracked or defective rotor blades by comparing fundamental frequencies obtained from periodic inspections can be used as a valuable supplement to the magnetic particle test. However, the frequency method in general would not be conclusive if used as the only method of detecting defective blading. Its maximum value is attained when used only as a supplement to the MPT to focus attention on defective blading that may otherwise go undetected.

REFERENCES

1. Leclerc, J. "A Bench for Excitation and Measurements for Blades of Compressors and Turbines." La Recherche Aeronautique, September, October 1959, pp. 43-45.

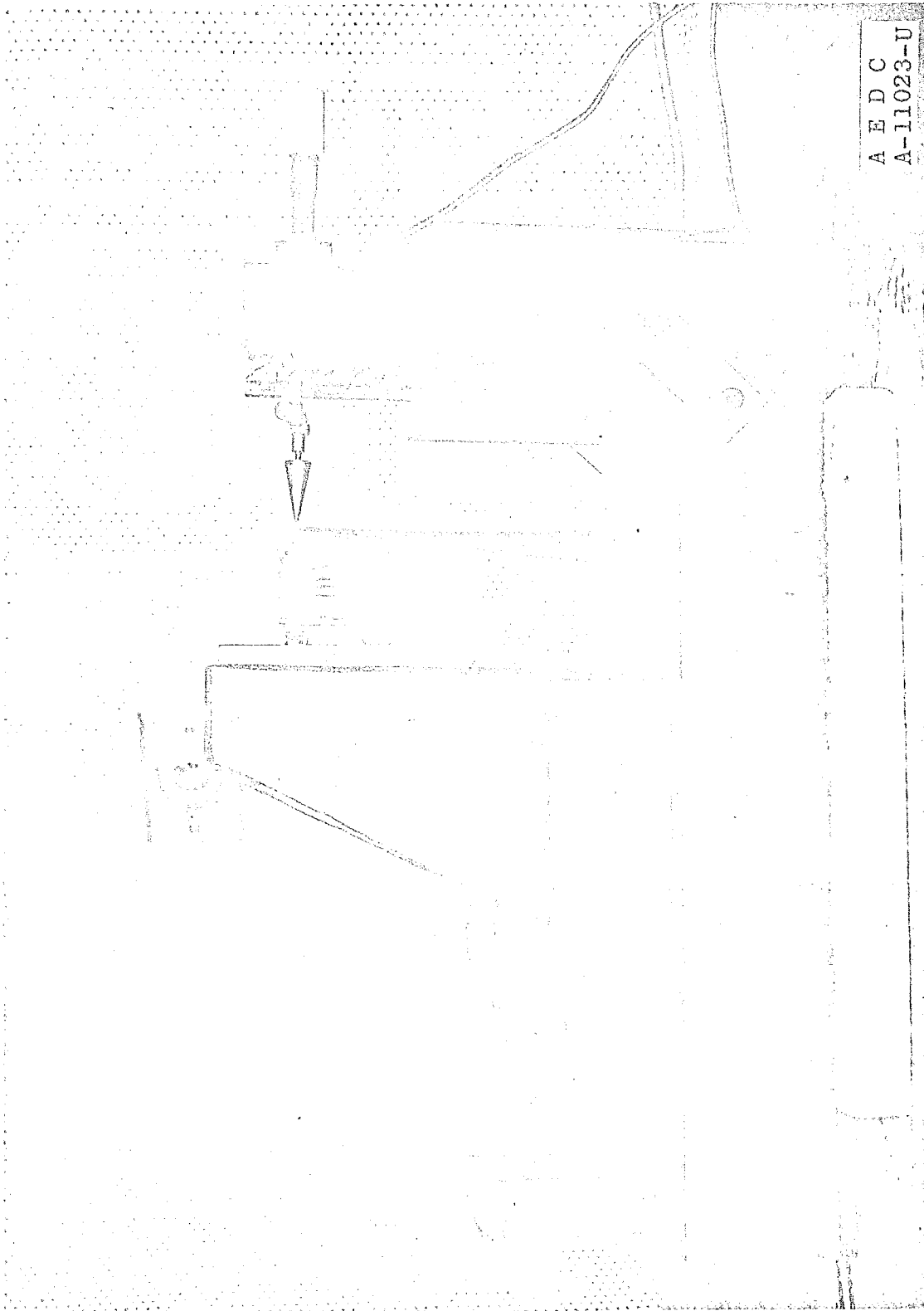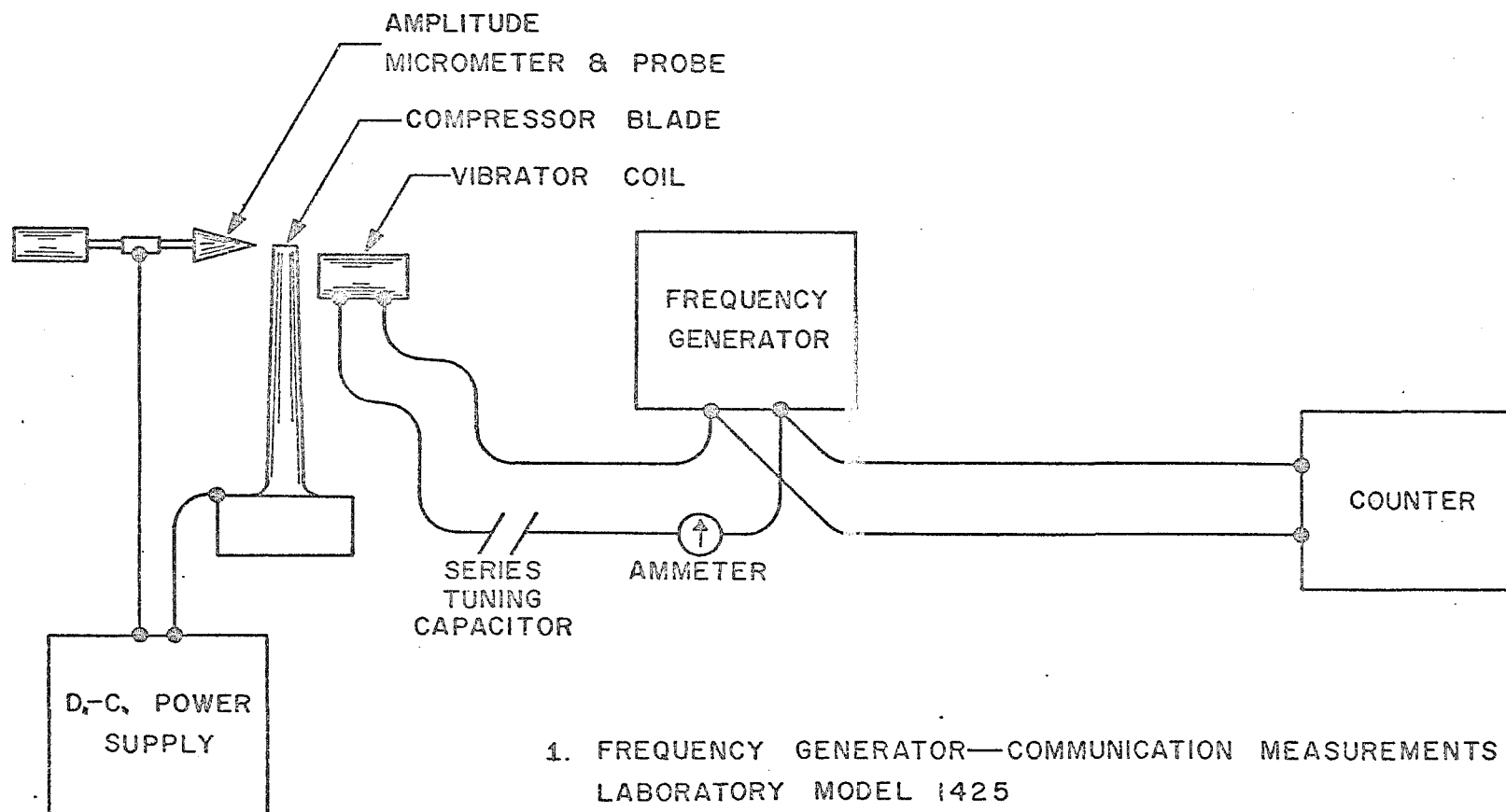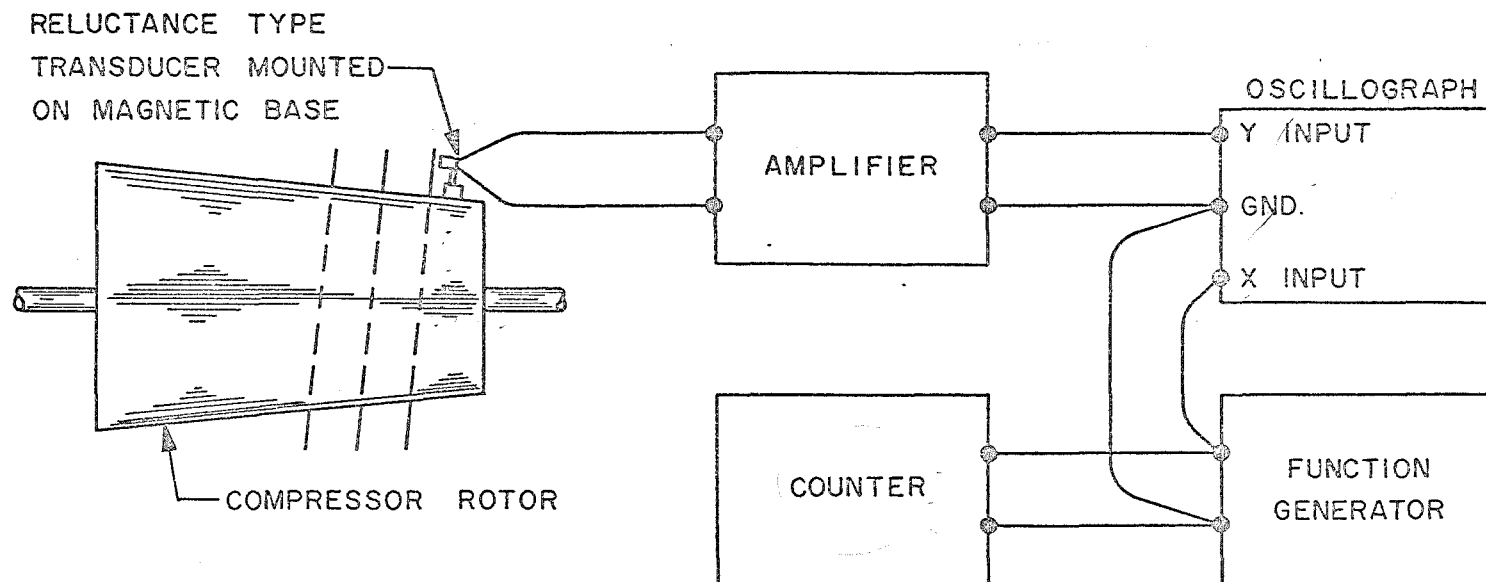2. McMaster, R. C. "Nondestructive Testing Handbook." The Ronald Press Company, New York, 1959.

AEDC
A-11023-U

Fig. 1 Blade Test Stand

AMPLITUDE
MICROMETER & PROBE

COMPRESSOR BLADE

VIBRATOR COIL

FREQUENCY
GENERATOR

COUNTER

SERIES
TUNING
CAPACITOR

AMMETER

D.-C. POWER
SUPPLY

1. FREQUENCY GENERATOR—COMMUNICATION MEASUREMENTS LABORATORY MODEL 1425

2. D-C. POWER SUPPLY—HEWLETT-PACKARD MODEL 712B

3. COUNTER—BERKELEY EPUT METER MODEL 554A

4. AMMETER—BURLINGTON MODEL 142

Fig. 2 Apparatus for Fatigue Testing Compressor Blades

RELUCTANCE TYPE
TRANSDUCER MOUNTED
ON MAGNETIC BASE

AMPLIFIER

OSCILLOGRAPH
Y INPUT
GND.
X INPUT

COUNTER

FUNCTION
GENERATOR

COMPRESSOR ROTOR

COMPONENTS

1. FUNCTION GENERATOR——HEWLETT-PACKARD MODEL 202A

2. COUNTER——BERKELEY EPUT METER MODEL 554A

3. AMPLIFIER——HEWLETT-PACKARD MODEL 450A

4. OSCILLOGRAPH——DUMONT TYPE 304A

Fig. 3 Apparatus Used to Determine Frequency of Compressor Blades
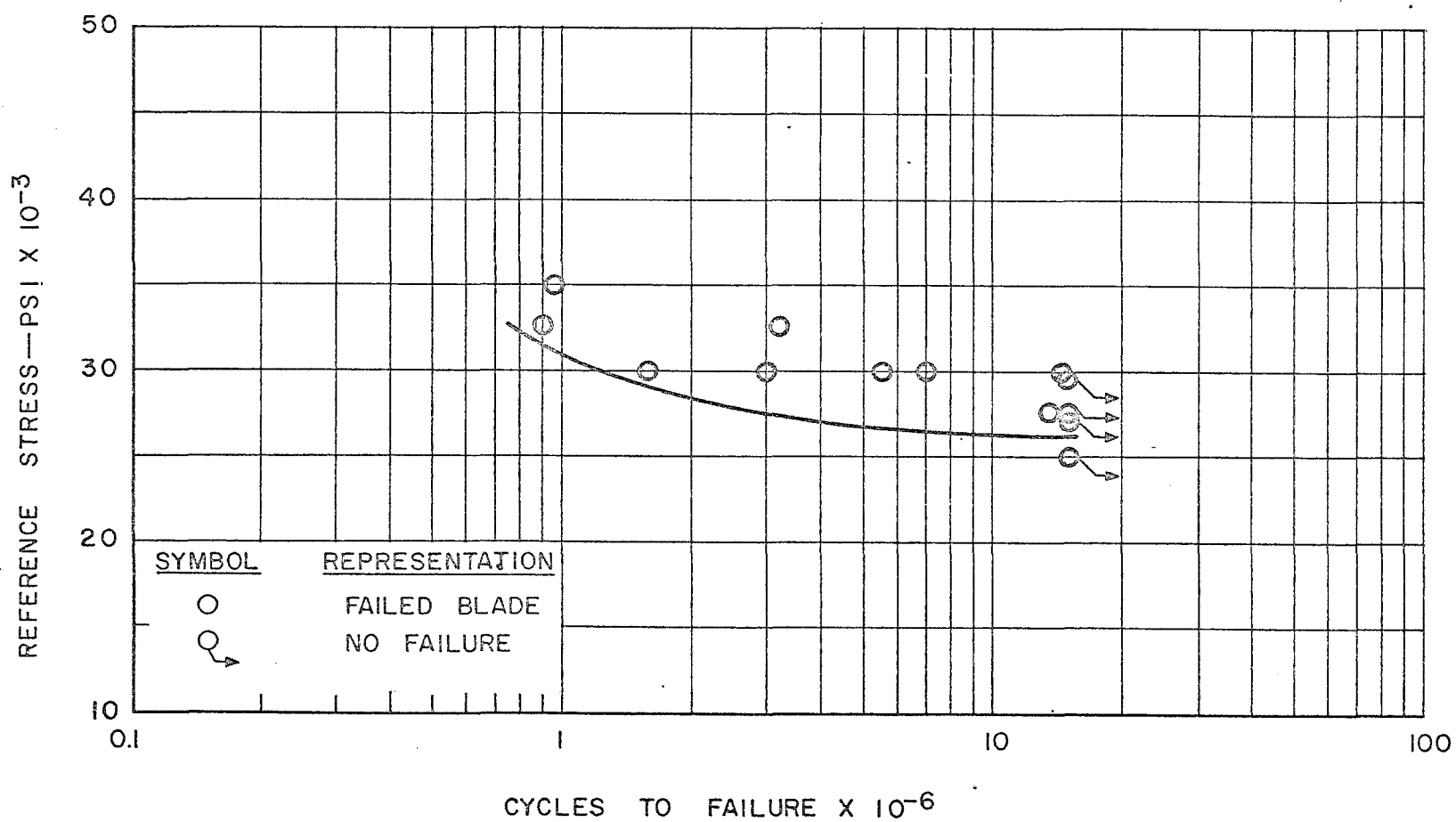
Fig. 4 Variation of Cycles to Failure under Completely Reversed Stress for Type-8630 Compressor Blades
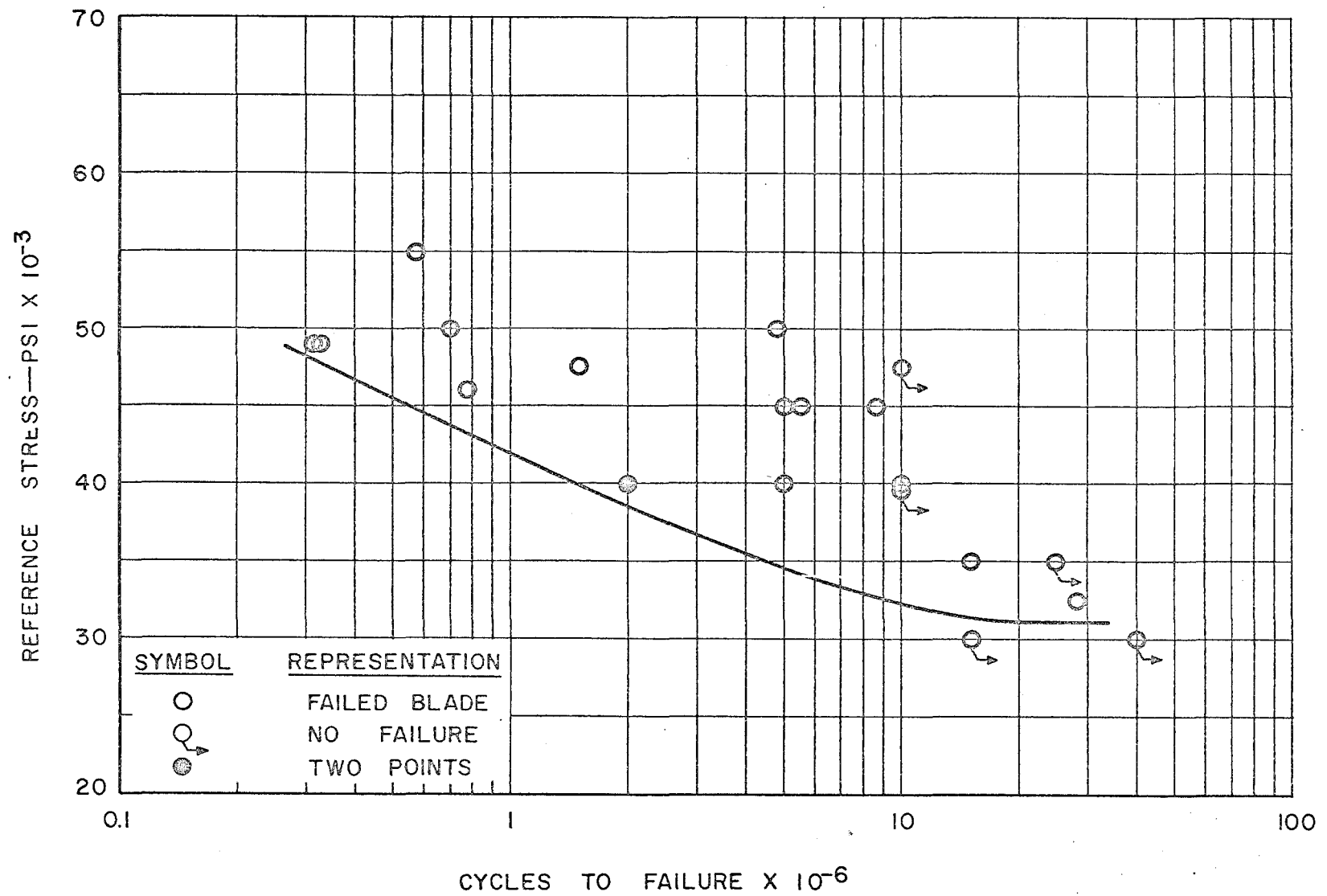
Fig. 5 Variation of Cycles to Failure under Completely Reversed Stress for Type-8630 Shot-Peened Compressor Blades
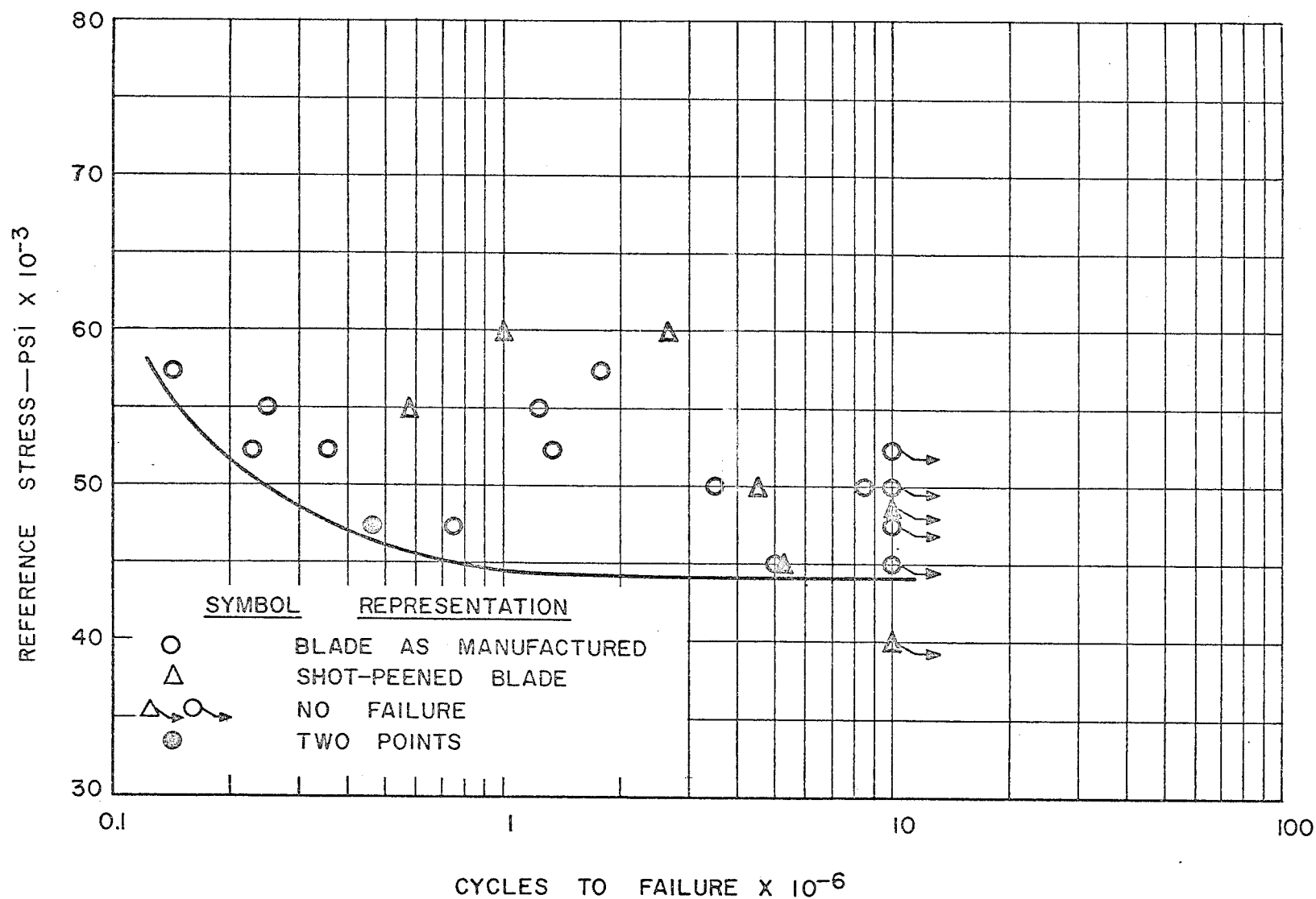
Fig. 6 Variation of Cycles to Failure Under Completely Reversed Stress for Type-403 Compressor Blades

IN COMPRESSOR HISTORY				
BLADE NUMBER	ROW NUMBER	FREQUENCY — CPS & DATE RECORDED		
		3—60	9—60	3—61
276	1	239.3	242.0	241.7
286	2	256.4	255.9	255.8
263 ✻	5	——	——	——
288	5	513.0	512.2	511.9
301	5	486.9	491.9	491.4

✻ REMOVED BY MPT—NO FREQUENCY RECORDED
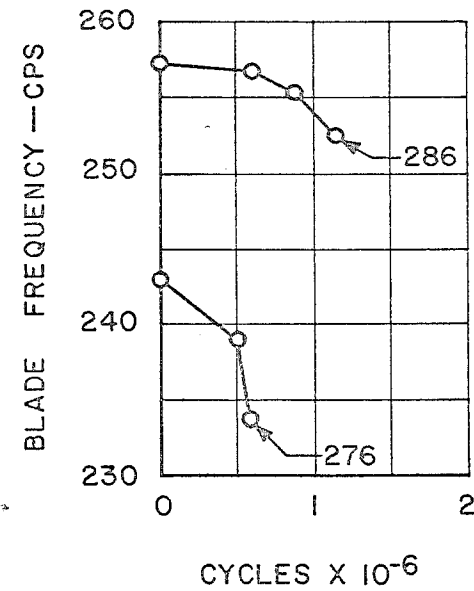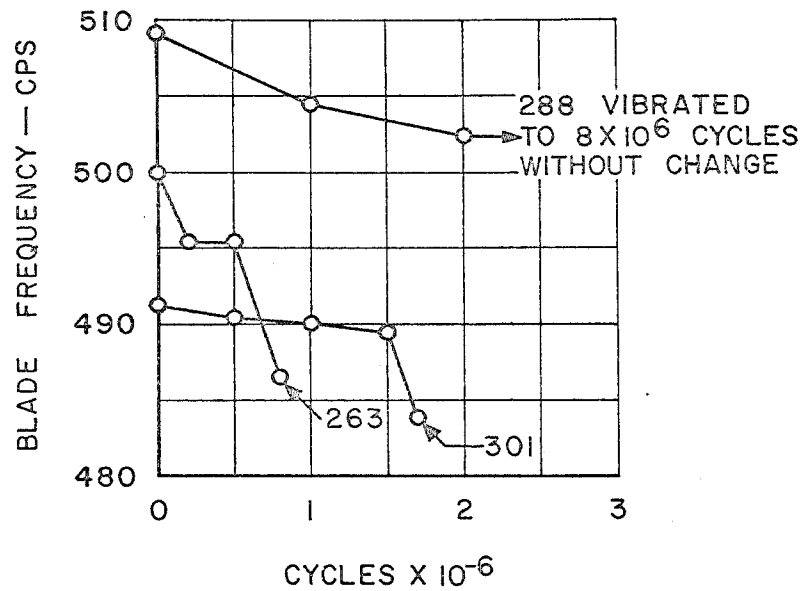
Fig. 7 Variation of Blade Frequency with Cycles of Vibration at 32,500-psi Stress

IN COMPRESSOR HISTORY				
BLADE NUMBER	ROW NUMBER	FREQUENCY — CPS & DATE RECORDED		
		3-60	9-60	3-61
311	9	912.7	921.9	920.7
324 ✳	8	775.4	775.2	775.4
297	9	906.7	906.7	906.0
295	9	886.2	885.5	884.8
285	8	——	934.3	904.9
283	8	766.5	772.8	766.1

✳ REMOVED BY MPT

Fig. 7 Concluded

24

COMPRESSOR 14
TYPE 8630 BLADE

COMPRESSOR 21
TYPE 403 BLADE

COMPRESSOR 22
TYPE 403 BLADE

LAST FREQUENCY MINUS PREVIOUS FREQUENCY—CPS

ROW NUMBER—50 BLADES PER ROW
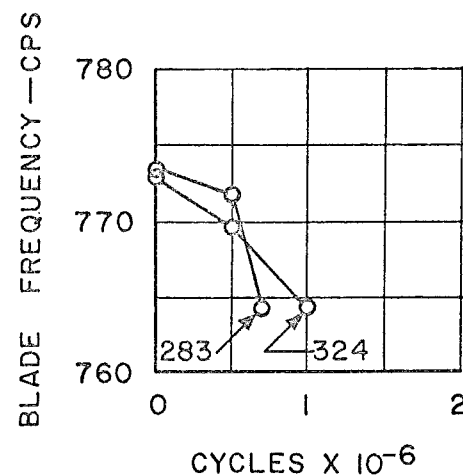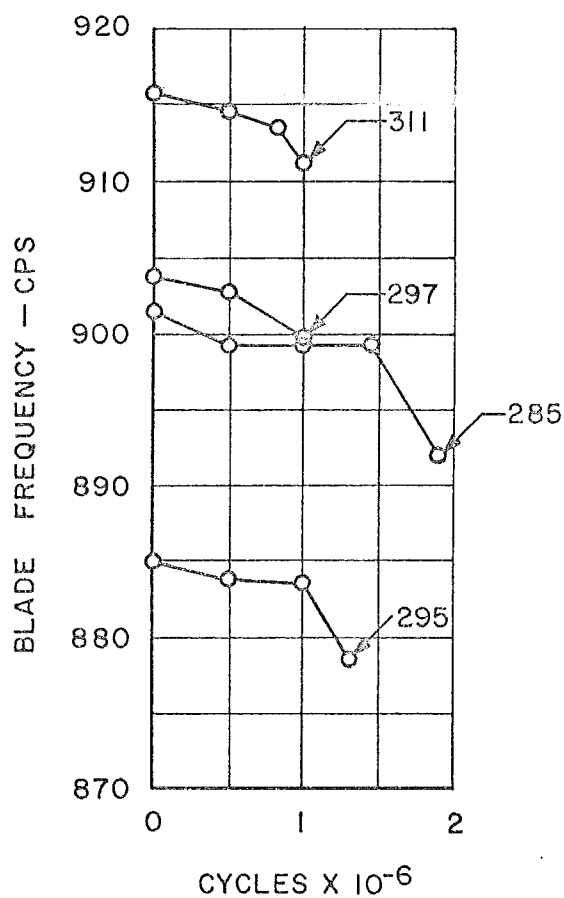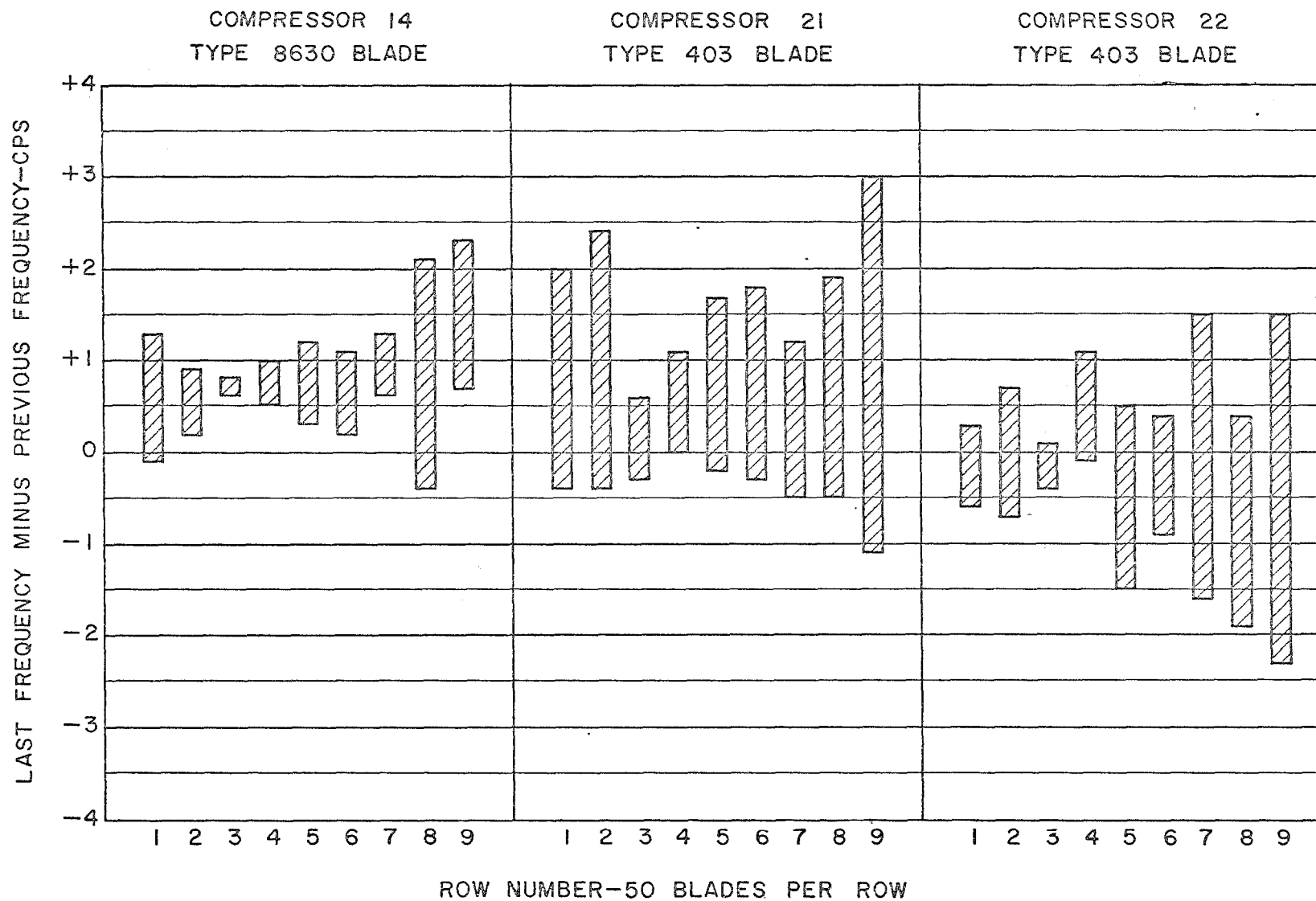
Fig. 8 Range of Differences between Two Rotor Blade Frequency Data

COMPRESSOR 11
TYPE 8630 BLADE

COMPRESSOR 12
TYPE 8630 BLADE

COMPRESSOR 13
TYPE 403 BLADE

ROW NUMBER—50 BLADES PER ROW

Fig. 8 Concluded

APPENDIX

BLADE TIP AMPLITUDE DETERMINATION

The following figures and tabulations were supplied by the Engineering Analysis Section, VKF, as supporting material for the calculation of blade tip amplitudes at a particular reference stress level. Computations not shown are on file in the above section.

Physical Constants of Types-8630 and 403 Blades

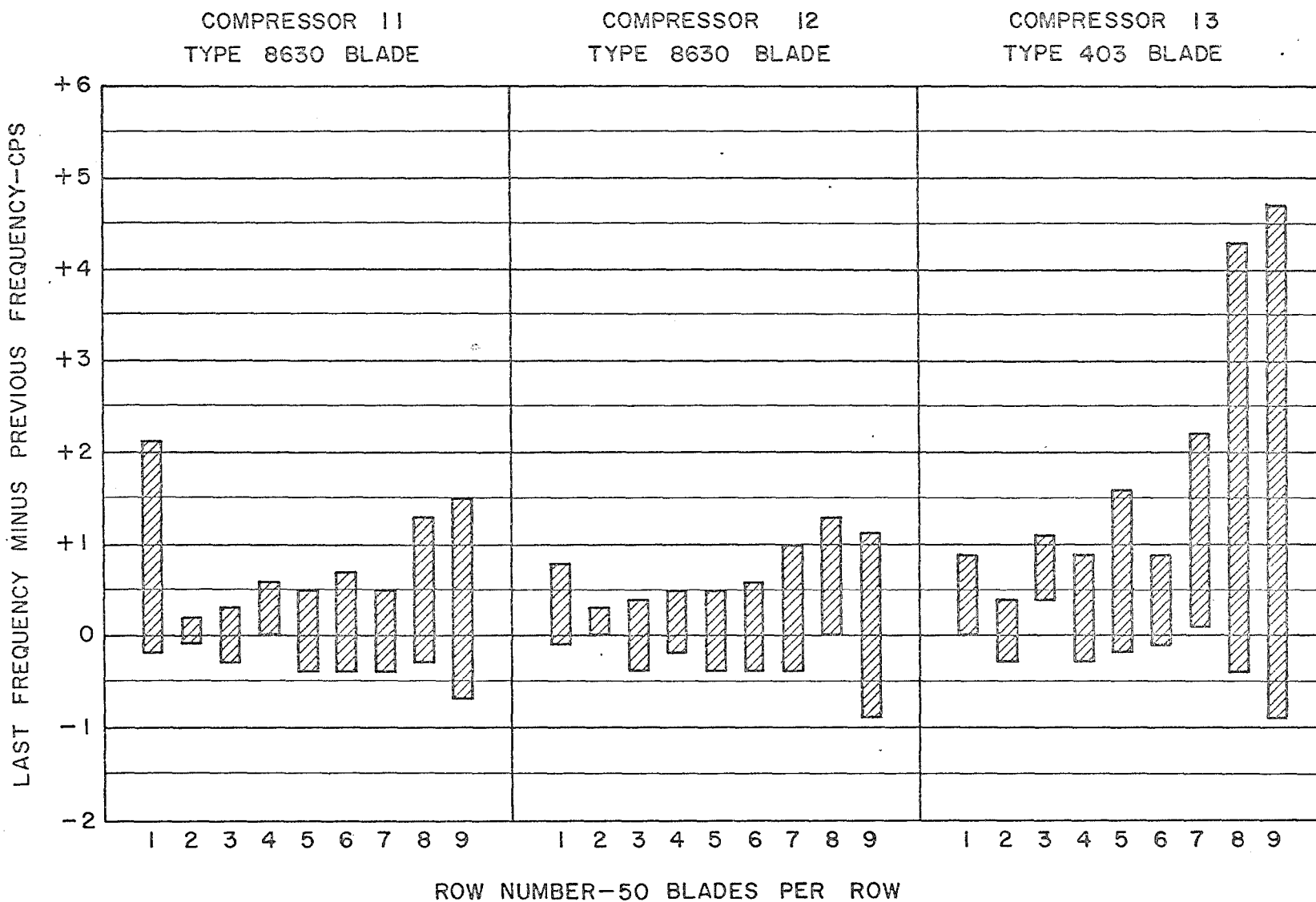| Blade Row Number	Type-8630 Cast Steel		Type-403 Forged Steel	
	Weight*, lb	Distance from Base to cg,* in.	Weight*, lb	Distance from Blade Base to cg,* in.
1	1.10	3.11	1.37	3.10
2	1.03	2.86	1.28	2.84
3	0.96	2.64	1.20	2.60
4	0.89	2.39	1.11	2.39
5	0.82	2.16	1.03	2.20
6	0.77	2.01	0.97	2.07
7	0.73	1.88	0.92	1.93
8	0.68	1.73	0.87	1.78
9	0.63	1.59	0.81	1.62

*Weight and distance to center of gravity (cg) measure for row 1 blade; others calculated assuming geometrical similarity except blade length

Using the above information, the tip deflection for a given reference stress was obtained by solving for Δt from:

$$\frac{\sigma_q}{\sigma_{1g}} = \frac{\Delta t}{\Delta_{1g}} \qquad \text{(Eq. 1)}$$

where Δt = deflection required at the tip of the blade to produce the reference stress

Δ_{1g} = deflection at the tip of a horizontally positioned blade resulting from dead weight of the blade, that is, a 1g load. See deflection versus length curve of Fig. 2-A.

σ_{1g} = stress at reference section resulting from a 1g load

σ_q = stress at reference section corresponding to Δt

The reference stress in each case was taken as the stress existing on a plane 1/4 in. above the base of the blade, that is, just above the fillet of the blade.

An example calculation for a row 1 type-8630 cast steel blade is as follows:

$$d = 3.11 - 0.25 = 2.86 \text{ in.} \left\{ \begin{array}{l} \text{the distance from} \\ \text{the blade cg to the} \\ \text{reference section} \end{array} \right.$$

$$M = wd = (1.10)(2.86) = 3.15 \text{ in.} - \text{lb}$$

where w is the weight of the blade above the reference section.

$$\sigma_q = \frac{MY_{max}}{I_{BB}} = \frac{(3.15)(0.235)}{0.008627} = 86 \text{ psi}$$

where Y_{max} and I_{BB} are taken from Table 1-A.

Solving for Δt from (Eq. 1) gives:

$$\Delta t = \frac{(0.000295)\sigma_q}{86} = 3.43 \; \sigma_q \times 10^{-6} \text{ in.} \qquad \text{(Eq. 2)}$$

For any stress value at the reference section of a row 1 type-8630 blade, Eq. (2) may be solved for the corresponding tip amplitude. Equations for each row and type of blade were determined similarly and are presented in the following tabulation:

Row Number	Δt	
	Type-8630 Cast Steel Blade	Type-403 Forged Steel Blade
1	$3.430 \; \sigma_q \times 10^{-6}$	$2.570 \; \sigma_q \times 10^{-6}$
2	$2.730 \; \sigma_q \times 10^{-6}$	$2.150 \; \sigma_q \times 10^{-6}$
3	$2.200 \; \sigma_q \times 10^{-6}$	$1.755 \; \sigma_q \times 10^{-6}$
4	$1.700 \; \sigma_q \times 10^{-6}$	$1.420 \; \sigma_q \times 10^{-6}$
5	$1.153 \; \sigma_q \times 10^{-6}$	$1.126 \; \sigma_q \times 10^{-6}$
6	$1.067 \; \sigma_q \times 10^{-6}$	$0.980 \; \sigma_q \times 10^{-6}$
7	$0.952 \; \sigma_q \times 10^{-6}$	$0.847 \; \sigma_q \times 10^{-6}$
8	$0.930 \; \sigma_q \times 10^{-6}$	$0.730 \; \sigma_q \times 10^{-6}$
9	$0.896 \; \sigma_q \times 10^{-6}$	$0.614 \; \sigma_q \times 10^{-6}$

TABLE 1-A

DETERMINATION OF AREA, MOMENT OF INERTIA, AND LOCATION OF THE X-X CENTROIDAL AXIS OF A TYPICAL ROW 1 ROTOR BLADE OF 8630 CAST STEEL

	Area	Distance	Moment of Area	Moment of Inertia	Moment of Inertia	Moment of Inertia
	A	$\overline{Y}$	$A\overline{Y}$	I_{cg}	Ad_1^2	Ad_2^2
Incr	in.^{2}	in.	in.^{3}	in.^{4}	in.^{4}	in.^{4}
1	.15375	.295	.0315	.00339	.03325	-
2	.10000	-.1333	-.0133	.000889	.06560	-
3	.4350	.2000	.0870	.02743	.09606	-
4	.5400	.3250	.1755	.05248	.06426	-
5	.6225	.4350	.2708	.0804	.03293	-
6	.6950	.5300	.3685	.11065	.01362	-
7	.7500	.6200	.4650	.1406	.001518	-
8	.7950	.6900	.5485	.1675	.000318	.018055
9	.8300	.7500	.6225	.1906	.005995	.006328
10	.8600	.8100	.6965	.2120	.01685	.000310
11	.8900	.8550	.7610	.2350	.03214	.000102
12	.9050	.9000	.8145	.2470	.04785	.003182
13	.9225	.9225	.8510	.2610	.05765	.006173
14	.9250	.9250	.8555	.2640	.06255	.006574
15	.9250	.9250	.8555	.2640	.06255	.006574
16	.9200	.9200	.8460	.2595	.05750	.005785
17	.9050	.9050	.8190	.2470	.0458	.003742
18	.8800	.8800	.7743	.2208	.0352	.001359
19	.8450	.8450	.7140	.2010	.01738	.000016
20	.8050	.8000	.6440	.17385	.01257	.001333
21	.7550	.7350	.5550	.1434	.002718	.008435
22	.695	.6500	.4518	.1119	.0001564	.025275
23	.6300	.5550	.3498	.0834	.005685	-
24	.5625	.4850	.2728	.05935	.02030	-
25	.4900	.3900	.2400	.0392	.03841	-
26	.4200	.2950	.1239	.0350	.06065	-
27	.3550	.1900	.0675	.01491	.0835	-
28	.3000	.0600	.0180	.0090	.1135	-
29	.2450	-.0600	-.0147	.004902	.1342	-
30	.2000	-.2000	-.0400	.002666	.1531	-
31	.0900	-.2400	-.0216	.000648	.0745	-
32	.0625	-.4500	-.0281	.000217	.07915	-
	19.50875		13.1617	3.864282	1.52749	.094323

Attachment Area
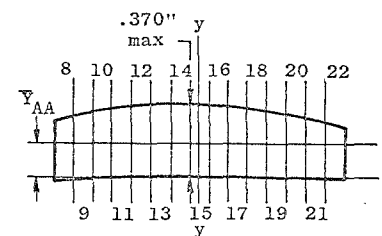Section A-A

Section B-B (Not to scale)
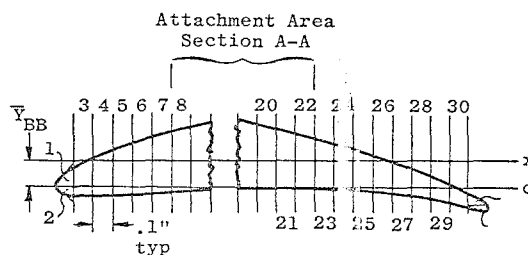
Section A-A (Not to scale)

$$A_{BB} = \frac{\Sigma_1^{32} A}{5^2} = \frac{19.50875}{25} = 0.78 \text{ in.}^2$$

$$\overline{Y}_{BB} = \frac{\Sigma_1^{32} A\overline{Y}}{5\Sigma_1^{32} A} = \frac{13.1617}{5 \times 19.50875} = 0.135 \text{ in.}$$

$$\therefore Y_{max} = 0.370 - 0.135 = 0.235 \text{ in.}$$

$$I_{BB} = \frac{\Sigma_1^{32} I_{cg} + \Sigma_1^{32} Ad_1^2}{5^4}$$

$$= \frac{3.864282 + 1.52749}{625} = 0.008627 \text{ in.}^4$$

$$A_{AA} = \frac{\Sigma_8^{22} A}{5^2} = \frac{12.8575}{25} = 0.514 \text{ in.}^2$$

$$\overline{Y}_{AA} = \frac{\Sigma_8^{22} A\overline{Y}}{\Sigma_8^{22} A \times 5} = \frac{10.8091}{12.8575 \times 5} = 0.168 \text{ in.}$$
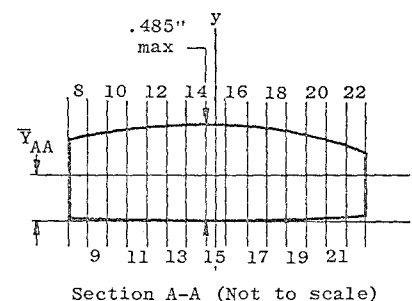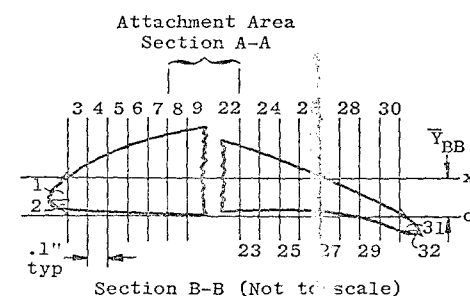
$$\therefore Y_{max} = 0.370 - 0.168 = 0.202 \text{ in.}$$

$$I_{AA} = \frac{\Sigma_8^{22} I_{cg} + \Sigma_8^{22} Ad_2^2}{5^4}$$

$$= \frac{3.1991 + .094323}{625} = 0.005269 \text{ in.}^4$$

All tabulated values
based on a 5x scale.

TABLE 2-A

DETERMINATION OF AREA, MOMENT OF INERTIA, AND LOCATION OF THE X-X CENTROIDAL AXIS OF A TYPICAL ROW 1 ROTOR BLADE OF 403 FORGED STAINLESS STEEL

	Area	Distance	Moment of Area	Moment of Inertia	Moment of Inertia	Moment of Inertia
	A	$\overline{Y}$	$A\overline{Y}$	I_{cg}	Ad_1^2	Ad_2^2
Incr	in.^{2}	in.	in.^{3}	in.^{4}	in.^{4}	in.^{4}
1	.205	.6133	.1257	.00775	.0251	-
2	.070	.2467	.0173	.00032	.0363	-
3	.520	.665	.3458	.04685	.0484	-
4	.672	.805	.5410	.10115	.0183	-
5	.793	.900	.7137	.16510	.0034	-
6	.893	.980	.8751	.23720	.00004	-
7	.970	1.039	1.0078	.30400	.0041	-
8	1.030	1.085	1.1176	.36470	.0246	.00744
9	1.084	1.130	1.2249	.42450	.0278	.00173
10	1.130	1.170	1.3221	.48100	.0452	.00000
11	1.164	1.190	1.3852	.52550	.0616	.06046
12	1.190	1.220	1.4518	.56175	.0744	.00298
13	1.205	1.245	1.4922	.58350	.0846	.00434
14	1.213	1.240	1.5041	.59475	.0917	.00594
15	1.212	1.245	1.5089	.59375	.0950	.00682
16	1.197	1.235	1.4783	.57200	.0873	.00506
17	1.167	1.215	1.4179	.53010	.0729	.00236
18	1.125	1.185	1.3331	.47450	.0544	.00025
19	1.075	1.160	1.2470	.41425	.0388	.00021
20	1.010	1.110	1.1211	.34330	.0193	.00364
21	.987	1.055	1.0413	.32050	.0071	.01305
22	.860	.985	.8471	.21200	.00009	.02943
23	.778	.915	.7119	.15700	.0019	-
24	.693	.820	.5683	.11090	.0128	-
25	.610	.730	.4453	.07565	.0323	-
26	.522	.615	.3210	.04742	.0970	-
27	.430	.490	.2107	.02650	.1349	-
28	.340	.340	.1156	.01310	.1732	-
29	.275	.165	.0454	.00694	.2150	-
30	.215	-.035	-.0075	.00331	.1574	-
31	.114	-.210	-.0239	.00076	.0404	-
32	.0225	-.375	-.0084	.00001	.0639	-
	24.7715		25.4874	8.30106	1.8545	.08371

Attachment Area
Section A-A

Section B-B (Not to scale)

.485" max

Section A-A (Not to scale)

$$A_{BB} = \Sigma_0^{32} A \times 5^{-2} = \frac{24.7715}{25} = 0.99 \text{ in.}^2$$

$$\overline{Y}_{BB} = \frac{\Sigma_0^{32} A\overline{Y}}{5\Sigma_0^{32} A} = \frac{25.4874}{5 \times 24.7715} = 0.206 \text{ in.}$$

$$C_{max} = 0.298 \text{ in. (measured)}$$

$$I_{BB} = \frac{\Sigma_0^{32} I_{cg} + \Sigma_0^{32} Ad_1^2}{5^4}$$

$$= \frac{8.30106 + 1.8545}{625} = 0.01625 \text{ in.}^4$$

$$A_{AA} = \Sigma_8^{22} A \times 5^{-2} = \frac{16.649}{25} = 0.666 \text{ in.}^2$$

$$\overline{Y}_{AA} = \frac{\Sigma_8^{22} A\overline{Y}}{5\Sigma_8^{22} A} = \frac{19.4826}{5 \times 16.649} = 0.234 \text{ in.}$$

$$C_{max} = 0.258 \text{ in. (measured)}$$

$$I_{AA} = \frac{\Sigma_8^{22} I_{cg} + \Sigma_8^{22} Ad_2^2}{5^4}$$

$$= \frac{6.9961 + .08371}{625} = 0.01133 \text{ in.}^4$$
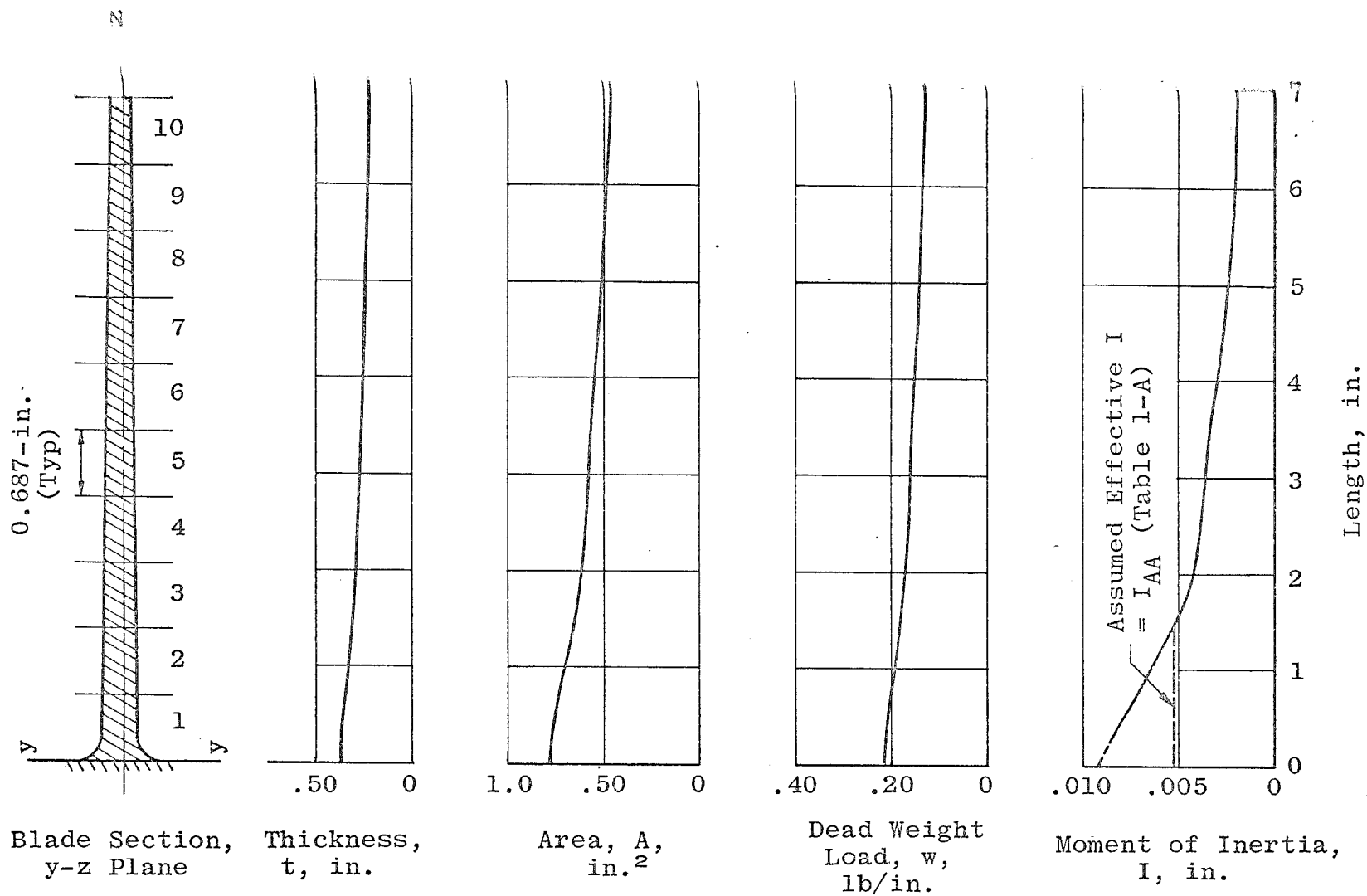
All tabulated values
based on a 5x scale.

Fig. 1-A Variation of Thickness, Area, Weight, and Moment of Inertia along a Typical Row 1 Rotor Blade of 8630 Cast Steel
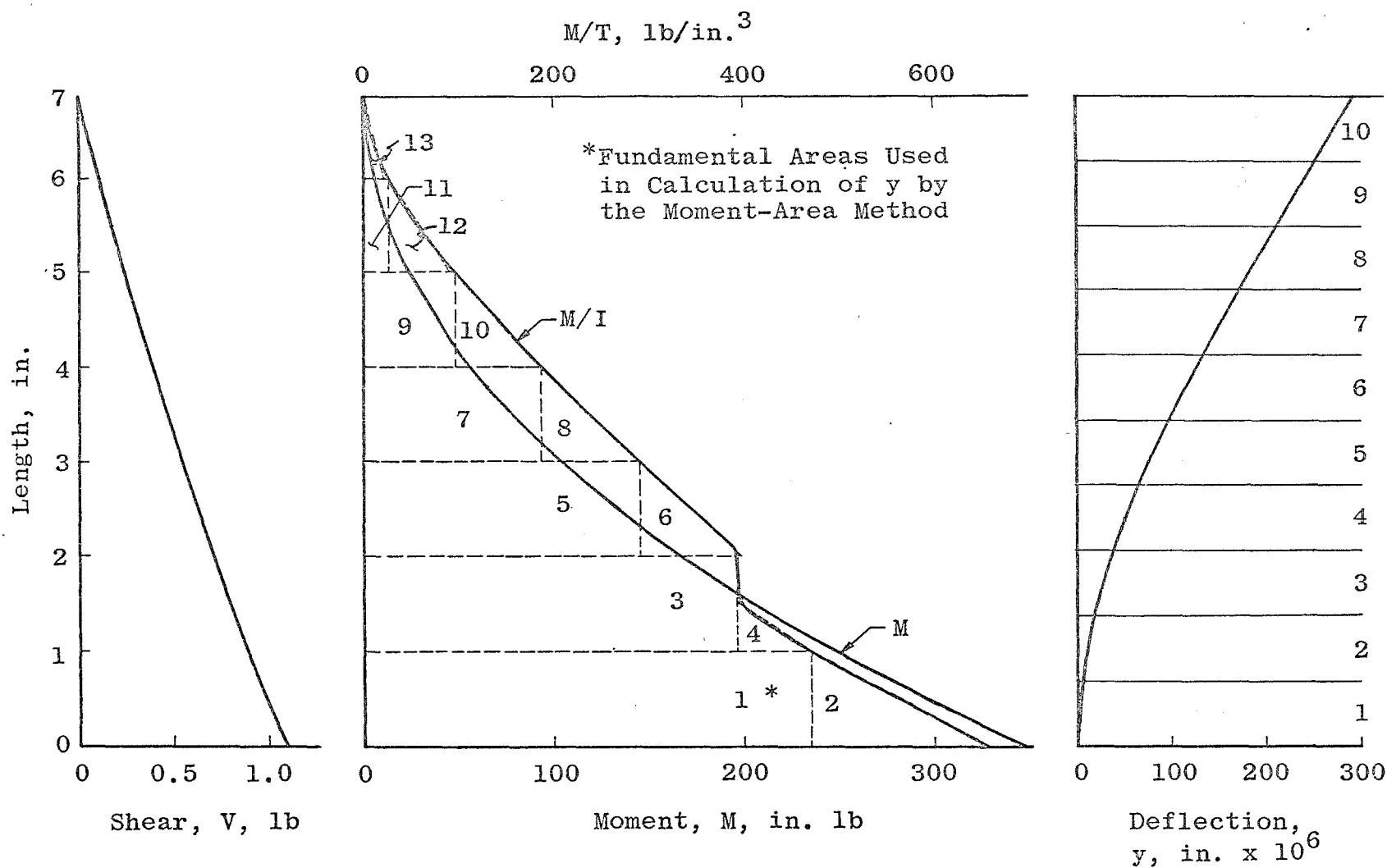
Fig. 2-A Shear, Moment, M/I, and Deflection along a Typical Row I Rotor Blade of 8630 Cast Steel

Blade Section   Thickness, t   x-Section Area,   Dead Weight   Moment of Inertia,
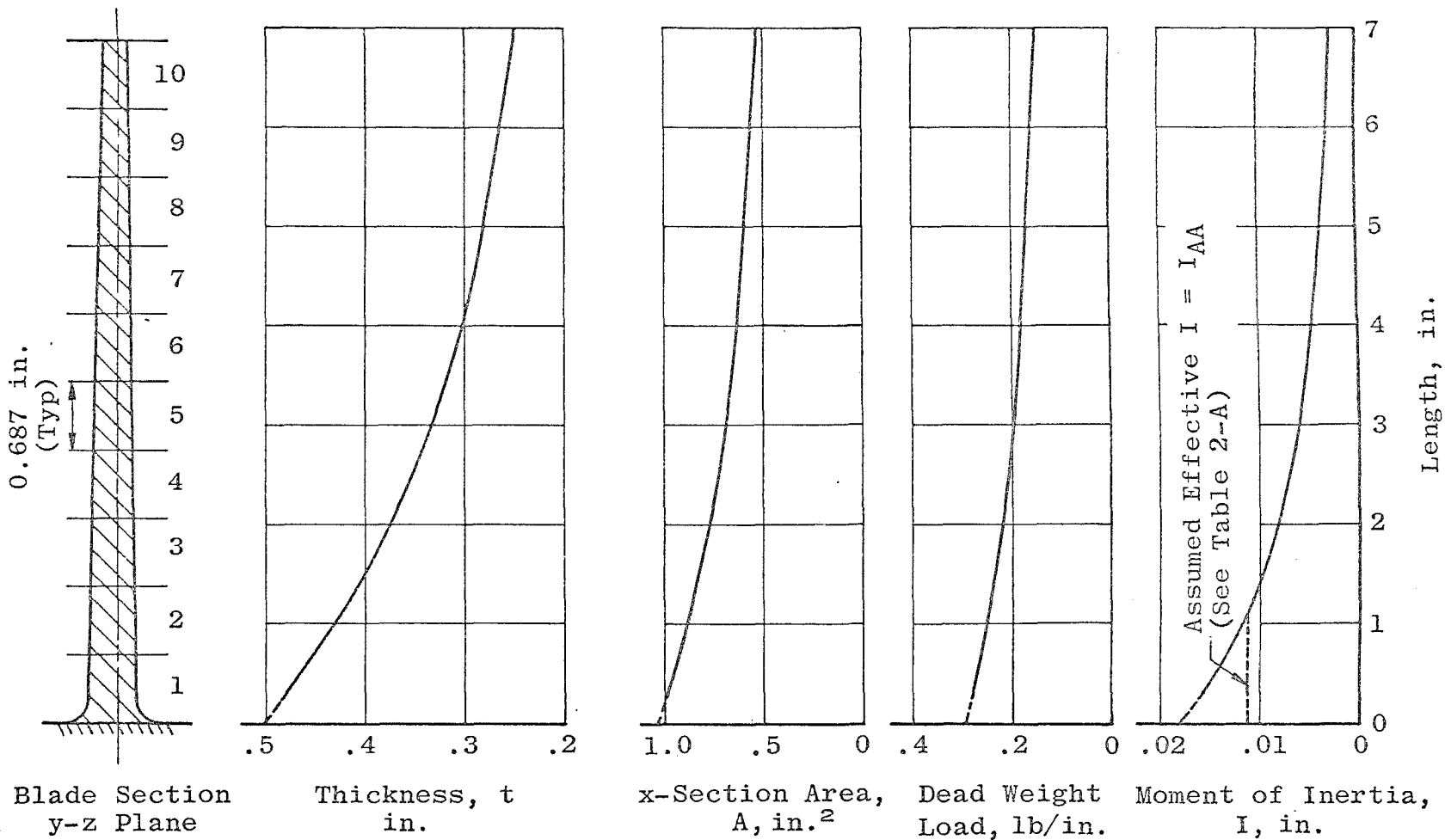y-z Plane   in.   A, in.²   Load, lb/in.   I, in.

Fig. 3-A Variation of Thickness, Area, Weight, and Moment of Inertia along a Typical Row 1 Rotor Blade of 403 Forged Stainless Steel
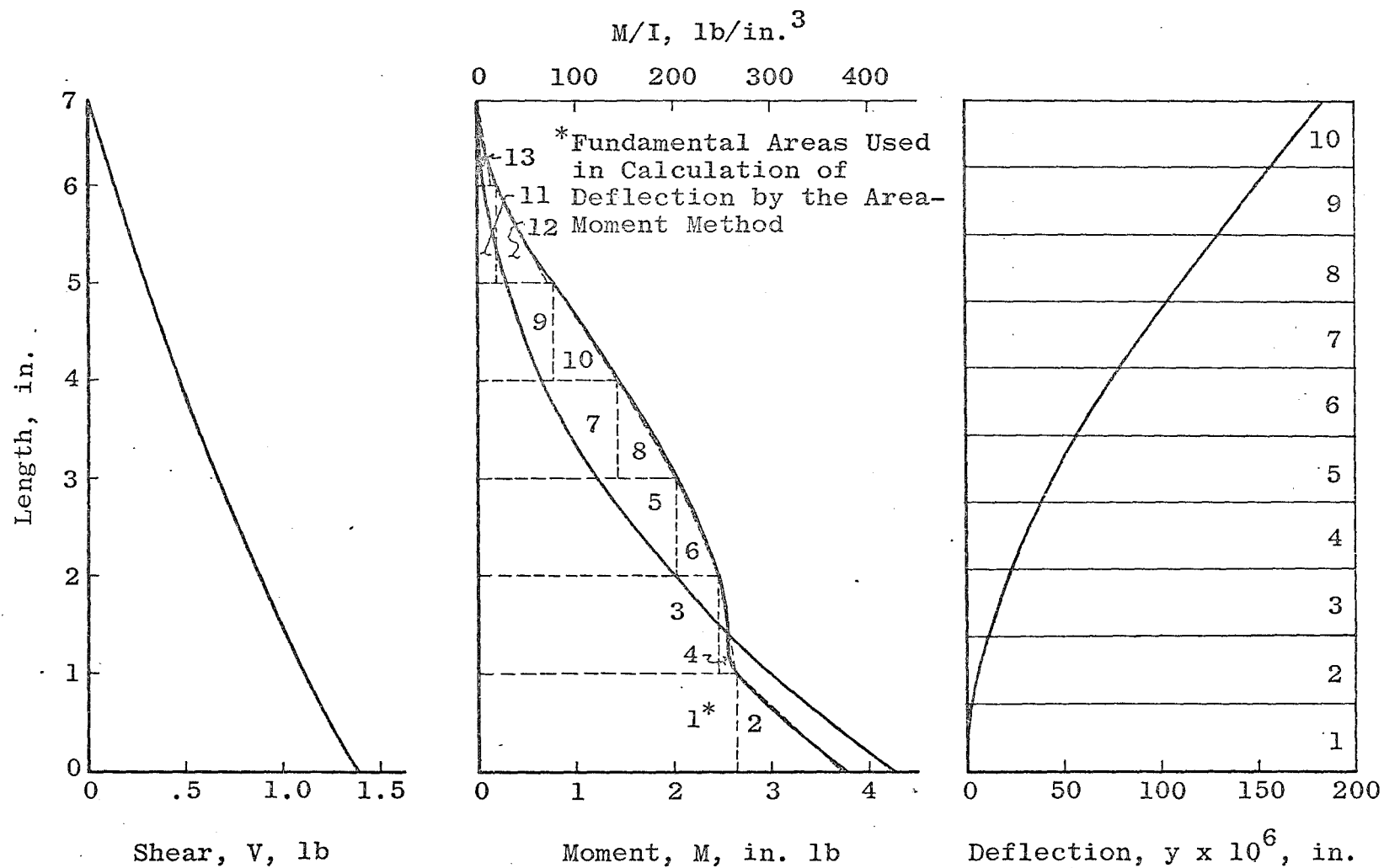
Fig. 4-A Shear, Moment, M/I, and Deflection along a Typical Row 1 Rotor Blade of 403 Forged Stainless Steel